Shaft CenterLINES

Extending machinery life

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This article summarizes basic rotor design considerations that are necessary to extend machine life. It outlines a variety of well-known, but often ignored, design principles which require further emphasis in future designs. The consequences of ignoring these principles are referenced to actual pump rotor failures. Where existing designs cannot be changed, recommendations for equipment monitoring are presented. Emphasis has been placed on the prevention of catastrophic failures due to shaft cracks on large vertical reactor coolant and recirculation pumps.

Introduction

For years rotors have been designed for the lowest cost consistent with machinery life specifications. In many instances, plant life extension programs have pushed machines to the limit and beyond the original design life goals. In redesigning rotors for longer life, application of basic design principles is as important as sophisticated analysis techniques.

Where machines must continue to operate for extended periods before redesigned rotors can be installed, a comprehensive monitoring system is required to warn of impending failures.

Nowhere are these principles more important than in nuclear reactor coolant and recirculation pumps. Research by the Bently Rotor Dynamics Research Corporation (BRDRC), involving sub-scale models and actual pump case histories, shows that there are generic pump design features that inherently limit life extension programs.

The application of basic mechanical engineering design principles can greatly increase machine life and reduce the possibility of shaft cracks leading to rotor failure.

Principle One:

Reduce rotor loading

Whenever possible, it is good design practice to reduce rotor loading. Reduced loads, of course, mean lower stresses, smaller rotor deflections, and lower bearing wear, all of which lead to longer rotor, seal, and bearing life.

Many large vertical pumps, such as reactor coolant and reactor recirculation pumps, employ a single volute discharge. This design causes a large side load that acts over a relatively small range, depending on pump flow and head. The heavy side load also improved rotor stability, at a time when instability mechanisms were not well understood. However, the higher side load also increases bending stresses which can readily cause propagation of shaft cracks initiated by the thermal stresses often seen in reactor coolant pumps.

A better basic design incorporates multiple volutes which distribute and balance the loads acting on the shaft, thereby reducing bending stresses. Balancing the loads increases the likelihood of bearing instability, but the mechanisms governing instabilities are now well-documented, and instabilities can be avoided through proper bearing design techniques.

Some pumps by design, others by maintenance practices, are assembled with a deliberate misalignment between the

pump and motor. This is done in order to preload sleeve bearings since it is well-known that increases in loading can delay the onset of instability. However, this unidirectional preload has the same detrimental effect, and it can add to any side loads already present.

Another loading concern is the sensitivity of the load change with changes in pump flow. High sensitivity can mean load increases more than 100% for relatively small (20-30%) changes in pump flow (Figure 1). A "flat" response is more forgiving of off design operation. At the very least, the designer should warn the operator what the consequences of operating off design will be in terms of life reduction. Many operators readily use the excess flow capacity of critical pumps without knowing the price paid in reduced rotor life.

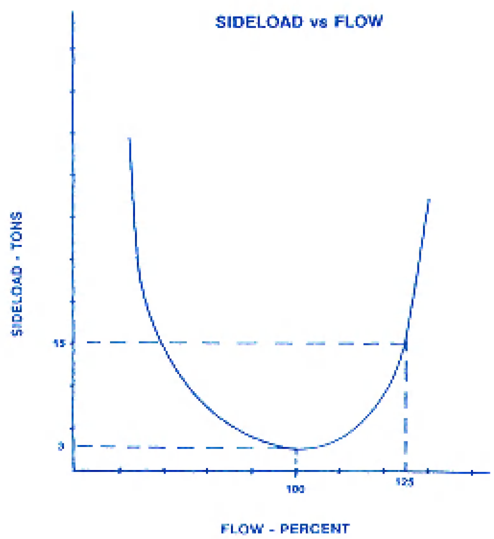


Figure 1
Typical pump side loading

Principle Two:

Reduce asymmetries

Physical shaft asymmetries such as keyways and holes for pins reduce the section and therefore the stiffness of the shaft in one direction. Under the influence of a radial side load, the shaft will bend more in the direction of the load, twice during each shaft revolution, causing cyclical alternating stresses which reduce the life of the shaft.

In order to reduce costs of manufacture, keyways are often all milled at the same angular location to permit all coupling, seal ring, impeller and other keyways to be cut with one machine setup. Lining up all the asymmetries compounds a bad situation.

In addition to creating alternating stresses, the twice per revolution (two times running speed) rotor response under side load can mask detection of rotor cracks, since the designed-in asymmetries and the asymmetry due to a crack can create the same response.

A return to the basic concept of design symmetry using multiple splines, instead of keyways and pins, will cost more initially but will inherently improve rotor life.

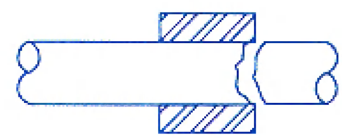
Principle Three:

Reduce stress concentrations

It is difficult to avoid stress concentrations in a built up rotor design. However, they can be minimized. Keyways and pins can be avoided by substituting multiple splines as noted above.

The problems of welding dissimilar metals and the effects of certain platings are well-documented and can be avoided. Crack initiation sites like these and areas where thermal stress cracking is possible should, as a minimum, be located away from high bending stresses if they cannot be avoided entirely.

Heavy shrink fits cause a phenomena that has only recently been documented. The shaft under a shrink fit stretches due to the tensile force caused by bending loads. A heavy shrink fit prevents the shaft from relaxing as the shaft rotates away from the bending load. The shaft is further stretched each time it bends. This ratcheting action eventually causes a cup shaped tensile fracture of the shaft (Figure 2).



- Usually said to be a "FRETTING CORROSION CRACK."
- Fretting is observable in Region of Shrink Fit.
- Looks just like a tension fracture of ductile material
  - Under **compression** due to a shrink fit, however, a **tension** crack should **not** occur, only a pure **shear** crack.

Figure 2
Cup shaped crack under shrink fit

Heavy shrink fits, usually applied to keep parts from rotating on the shaft or moving axially, should be replaced with multiple splines or radial keyways and positive axial retention features such as spanner nuts.

Threads, grooves and step changes in shaft diameter cannot be avoided but can be located away from high stresses and other stress concentration features.

A particularly bad combination is a stress concentration under a heavy shrink fit. Two common pump designs combine these problems. One has a step change and groove near a shrink fit; the other places an anti-rotation pin under a shrunk-on collar (Figure 3).

The best time to minimize stress concentrations is at the preliminary design stage, when first determining the layout of the rotor assembly.

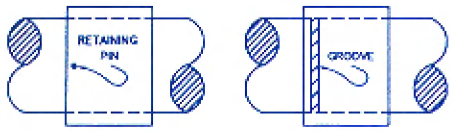


Figure 3
Combined stress concentrations

Principle Four:

Avoid operation near half of a resonance

The hazards of operating rotating equipment near a resonance frequency (critical speed) have of course been well-known for over a century. However, the effects of higher order resonances on rotor failures have only recently been documented.

It is not uncommon to find that actual machine resonances are very different from calculated theoretical values. This is often caused when actual bearing stiffness and the stiffening effects of seals, wear rings and the pumped fluid itself are either ignored or are not known. In addition, it is necessary to calculate resonances based on the entire rotor system of pump, motor, coupling and supports, rather than to simply isolate the pump rotor for analysis.

In order to avoid resonance problems, many end-users specify that all resonances must be well above operating speed. In fact, most reactor coolant and recirculation pumps operate at speeds below the first resonance. This takes care of the classical problems of operation at a lateral resonance but ignores higher resonances, especially the very serious consequences of a lateral resonance near twice running speed.

A crack reduces the shaft stiffness in one direction in much the same way as the asymmetries noted above. A side load acting on a cracked rotor will cause more bending in the direction of the load twice per revolution (Figure 4). If the rotor has a resonance near twice operating speed, the deflection response of the shaft will be amplified (Figure 5) accelerating crack propagation and leading to failure of the shaft.

Torsional resonances of the rotor system are often ignored. The same cautions should be taken with torsional resonances as are taken with lateral resonances.

Simple mode shape information, available from resonance speed calculations, is a basic design tool. The relative shaft deflections and node locations are valuable information when trying to place seals, couplings and instrumentation along a rotor system (Figure 6).

The four general principles listed above are certainly not new; they are taught at the undergraduate level. It is important to review these simple principles prior to new rotor design and redesign of existing rotors. Adherence to sound basic design rules, based on long experience and common sense, can have more effect on increasing machine life than exotic materials and intricate analysis techniques.

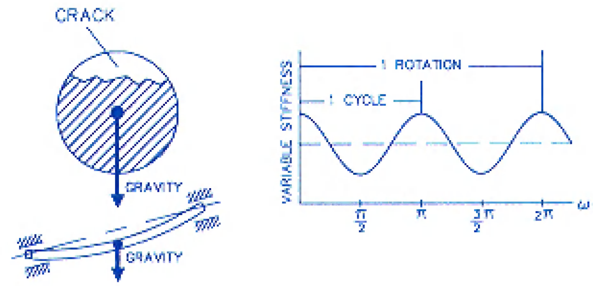


Figure 4
Vibrations caused by steady radial load and crack

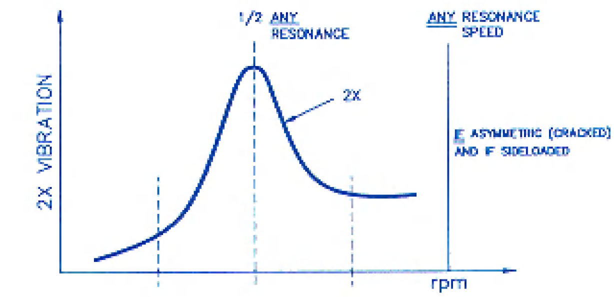


Figure 5
Example of half resonance response

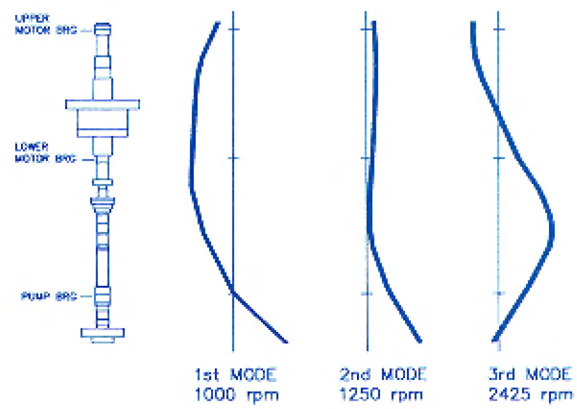


Figure 6
Typical pump/motor mode shapes

Vibration monitoring for failure prevention

It can take many years to replace an existing design with an improved rotor. Failures must be detected in time to plan maintenance and repairs and well in advance of catastrophic rotor failures. New rotors, costing hundreds of thousands of dollars and in critical service, must also be protected. Fortunately, cost-effective monitoring systems are available that can detect rotor problems at an early stage.

The tools are available to detect and diagnose rotor faults in time to prevent serious rotor damage. Proper vibration monitoring and analysis can detect rubs, imbalance, insta-

bilities, loose parts, bearing problems, misalignment, structural faults and the most serious rotor fault, the shaft crack. The best systems are comprehensive, looking at all the available data including vibration amplitude, phase and frequency. This data is used for automatic machine shutdown, for diagnostics and to build machinery trend histories.

A lot of progress has been made in detecting shaft cracks. Cracks as small as 15% of shaft diameter have been detected. Machines have been saved months before catastrophic rotor failure. The best tools for shaft crack detection are eddy current shaft displacement transducers, a Keyphasor® transducer and a monitoring/computer system that can display amplitude, phase and frequency data.

Too many vibration monitoring systems fail to make use of all the available data and analysis techniques by ignoring vibration phase data. Amplitude and frequency data are important, but often the early warning of a crack is provided by phase data. In a recent case of a save involving a cracked reactor coolant pump shaft, vibration phase angle changes were seen months before amplitude and frequency changes.

Proximity probes, mounted to directly observe shaft movement, are preferred over seismic transducers mounted on housings or support casings. Rotor response, not case response, must be known to detect shaft cracks.

There are many excellent sources which explain the vibration symptoms associated with shaft cracks and the analysis techniques used to examine rotor vibration data. Briefly these symptoms may be summarized as follows:

- Changes observed in the synchronous (1X) shaft speed relative vibration amplitude and/or phase angle response, beyond the variations seen during normal machine operation, are a good indication that physical changes have occurred in the rotor system. These changes are best seen on a monitor that displays 1X amplitude and phase and, if recorded, can be analyzed using Amplitude and Phase versus Time (APHT) plots. An on-line computer system is an invaluable tool which stores this data and automatically produces the APHT plots. The computer can also be "taught" to recognize changes that fall outside a previously defined Acceptance Region (Figure 7).
- Changes observed in the twice rotative (2X) component of vibration, amplitude and/or phase, beyond normal variations, are an early symptom of shaft cracks in many pumps. A crack, except for the rare full annular crack, reduces shaft stiffness in one direction. During each rotation of the shaft the stiffness varies twice. If a radial side load is present, the shaft responds with two deflections per revolution, i.e. dynamic motion at 2X rotative speed. Unfortunately, large designed-in asymmetries such as keyways also produce this 2X response and can mask the early 2X amplitude, but usually not the phase changes, due to a crack (Figure 8).

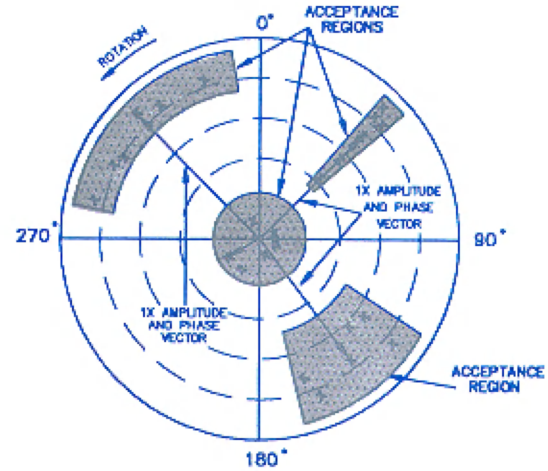


Figure 7
Vibration vector Acceptance Regions

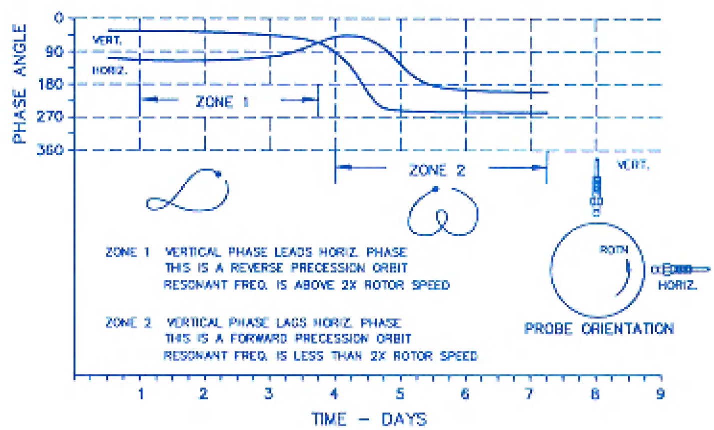


Figure 8
Twice synchronous phase angle vs time

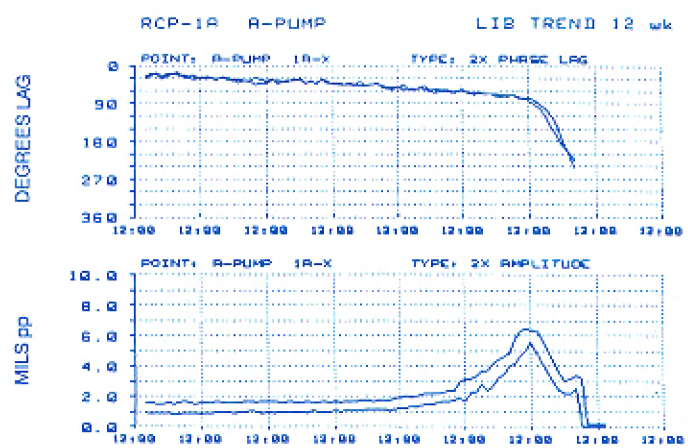


Figure 9
APHT - Critical speed response

- The same monitors and plots appropriate for synchronous (1X) vibrations are also used for twice synchronous (2X) vibrations. Note that the decrease in shaft stiffness associated with a crack will lower the resonance of the system. A resonance above twice running speed or above running speed may reduce in frequency due to reduced shaft stiffness. If the resonance frequency drops down into twice running speed range, the 2X response will be greatly amplified. This will appear on an APHT plot and will resemble a classical Bodé (amplitude/phase versus speed) critical speed plot with no change in rotative speed (Figure 9).
- Erratic changes in 1X and or 2X amplitude and/or phase during transient operation are indicative of shaft cracks. Analysis of data taken during startups and coastdowns displayed in Polar (Nyquist) plot format is one of the best techniques used to discover cracks. Another useful format is the cascade frequency plot which can be examined for peaks in the 2X (and to some extent, 3X and 4X) components that coincide with a known system resonance (Figures 10 and 11).
- An erratic 1X response to attempted balancing is a good indication of the nonlinear or changing behavior of a rotor. Again, this is best seen in Polar plot format.

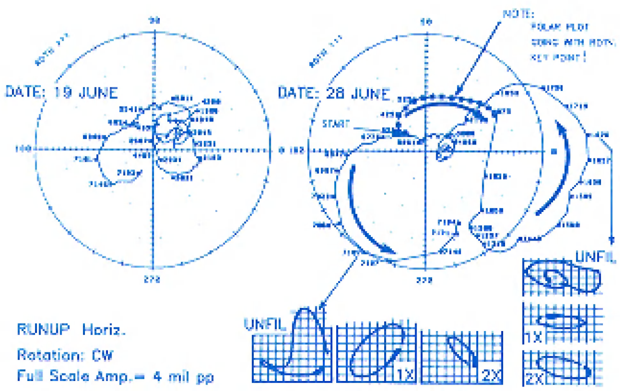


Figure 10
Cracked shaft - polar plot

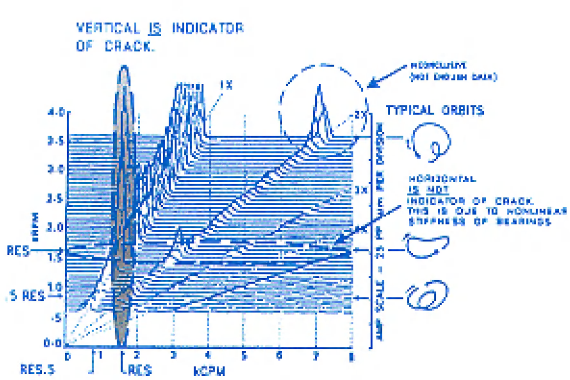


Figure 11
Cracked shaft- spectrum cascade

- Multiple Orbit plots should be used to study the rotor's orbital shape either at strategic points in the transient period or at different times. Typically, internal loops may be observed when the 2X vibration excites a resonance of the rotor system (Figures 5, 8, 10 and 11).

The designer can enhance rotor vibration analyses by eliminating asymmetries, by locating probes away from no-motion lateral node points, and by providing the user with characteristic machine information such as resonances, mode shapes and accurate machine drawings. The machine operator should be provided with information on how machine loading affects rotor life, and the designer must also consider the use of a comprehensive vibration monitoring system during the preliminary design stages. Provisions for a comprehensive system must be included to increase machine protection and to enhance machinery diagnostics.

Conclusions:

Adhere to simple basic design principles: minimize loading, reduce asymmetries, reduce stress concentrations, and avoid resonances, to provide a strong basis for increasing rotor life.

Modern comprehensive vibration monitoring systems can provide early warning of rotor problems in time to prevent failures.

The designer needs to know the basics of vibration monitoring systems and analysis techniques so the rotor design enhances monitoring and early fault detection.

The designer must provide the end-user with the information (drawings, resonances, mode shapes, loading, etc.) needed to aid in diagnostics and to avoid operation that is detrimental to machine life.

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